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Four-Square Gear Rig (FSG) Equipment Capabilities

by Kyle Olchewsky, Mark Riggs, and Adrian Hood

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Kyle Olchewsky
SURVICE Engineering

Mark Riggs and Adrian Hood
DEVCOM Army Research Laboratory

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ORCIDs: Kyle Olchewsky, 0009-0009-9116-8776; Mark Riggs, 0000-0003-1807-5994; Adrian Hood, 0000-0003-4044-0302

14. ABSTRACT

ARL is home to the Four-Square Gear Rig (FSG), a high-speed spur gear rig designed for component-level gear and transmission research. The FSG enables U.S. Army, academic, and industry partners to engage and explore new gear technologies to increase power density, survivability, and reliability for geared systems. The Army will use the FSG to characterize gear and transmission technologies such as materials, coatings, additive manufacturing methods, diagnostic sensors, machine-learning–based diagnostic algorithms, and improved lubrication methods. The FSG's four-square gear arrangement mechanically recirculates power though the test gears, which have a 3.5-inch pitch diameter and 1:1 ratio, to provide a shaft power of up to 790 hp at the test gear mesh with a maximum speed of 12,500 rpm. The FSG is an Army-unique testbed because it is designed for highly controlled loss-of-lubrication experiments and debris injection within the gear mesh. The purpose of this report is to record the FSG's equipment capabilities to inform current and future gear technology research collaborations.

15. SUBJECT TERMS
Mechanical Sciences, gears, spur gears, gearbox, transmission, powertrain, lubrication methods, additive manufacturing, loss-of-lubrication

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Kyle Olchewsky	(520) 852-2945

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1. Introduction

The Four-Square Gear Rig (FSG), shown in Figure 1, is a newly fabricated research rig that was designed for component-level analysis of gear technologies. FSGs have been instrumental in gear and lubrication research and development for decades. The FSG bridges the gap between current coupon-level research capability of the high-speed ball-on-disc tribometer (Wedeven Associates Machine)[1] and the system-level experimental capabilities provided by the Vehicle Innovative Powertrain Experimental Research (VIPER) testbed.[2] The new research capabilities enable investigation and assessment of novel gear technologies in a controlled environment with repeatable conditions.

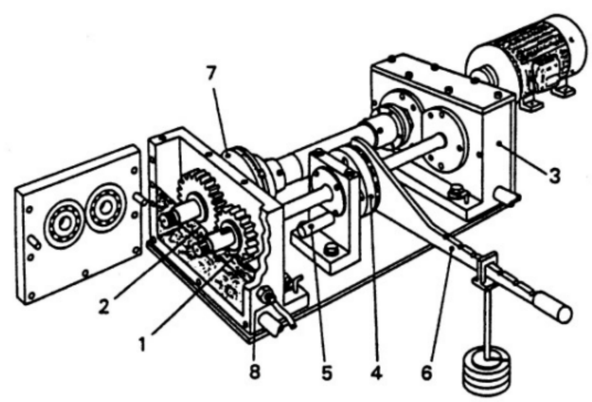
Figure 1. FSG.

The FSG's component-level research capability will be used to characterize gear coatings, gear materials, oil additives, and surface finishes through a variety of experimental protocols. These protocols include load-carrying capacity, endurance, debris injection, and loss of lubrication. Conditions are closely controlled and monitored through the FSG's integrated data acquisition system. Characterization of the data, including power transmission performance and postprocessing gear-tooth surface analysis, will provide new knowledge for gear technologies. FSG research and development will inform system-level experiments to support technology transitions to the Soldier.

2. Technical Specifications

2.1 Four-Square Principle and FSG Overview

The FSG gets its name from the basic four-square torque regenerative loop it uses to produce high torques on the test gears with small input power. Figure 2 depicts a traditional FSG machine. The four-square arrangement consists primarily of two reaction gears and two test gears. The speed is controlled by a motor, and the torque is controlled by a torque application component. The torque application component can be a mechanical lever arm to lock in the torque before operation, or a dynamic control such as a hydraulic torque actuator. The resulting power is circulated between the reaction gearbox and the test gearbox. Therefore, the motor is only required to provide enough torque to overcome the frictional losses in the system. Thus, the four-square design can use a significantly smaller motor than a traditional motor and dynamometer setup to produce the same power levels at the test gears.



1. Test gear
2. Test pinion
3. Reaction gearbox
4. Rotating coupling
5. Locking pin
6. Torque applicator
7. Torque meter
8. Temperature sensor

Figure 2. Traditional FSG machine with static loading.[3]

The FSG comprises a self-contained gear machine, an electrical cabinet, and a control computer. The machine skid houses all mechanical equipment required to run experiments including the motor, hydraulic torque actuator, oil reservoirs, pumps, heaters, heat exchangers, and filtration (Figure 3). The electrical cabinet contains the controller and drives. The control computer allows the operator to run experimental protocols through scripts that automatically adjust the speed, torque, lubrication temperature, and oil flow rate during experiments.

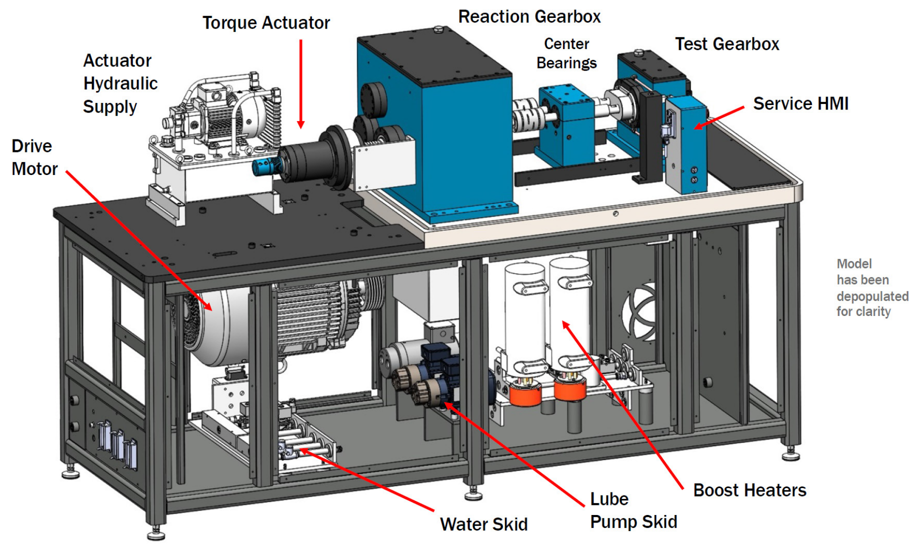


Figure 3. FSG schematic showing mechanical components including motor and heater locations.

2.2 FSG Technical Specifications

The FSG uses a 50-hp electric motor to produce 790 hp at the test gears with speeds up to 12,500 rpm. The input motor is rated for 3,600 rpm and uses a pulley and reaction gearbox to produce 12,500 rpm at the test gears. The hydraulic torque actuator is rated up to 332 ft-lb of torque and can be controlled dynamically at all operating speeds. Independent speed and torque controls allow for a vast range of power conditions up to a maximum of 790 hp at the test gears. A schematic of the FSG's power loop components is shown in Figure 4. Technical specifications of the FSG are in Appendix A.

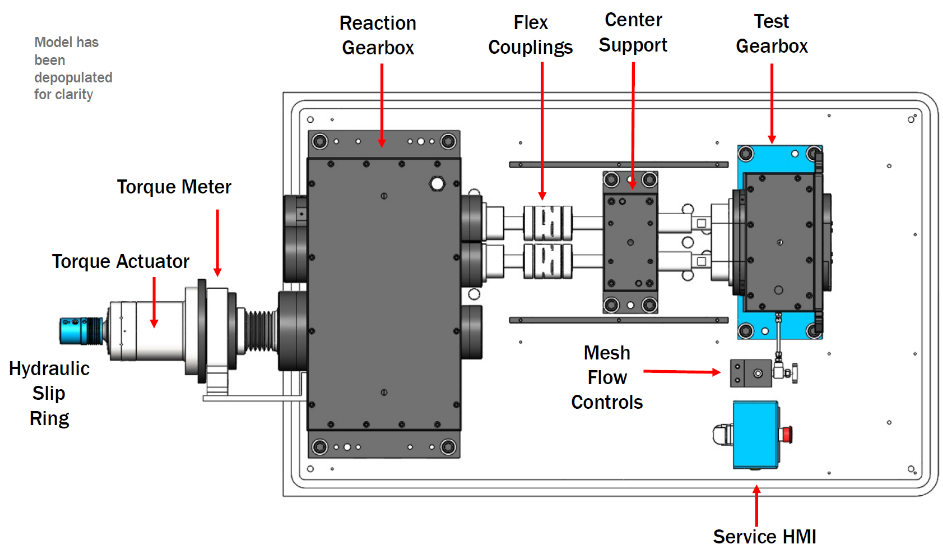


Figure 4. FSG power loop showing location of hydraulic torque actuator and torque meter.

The reaction gearbox is a speed-increasing gearbox that increases the output from the electric motor to the full test gear speed of 12,500 rpm. It is lubricated with a separate oil reservoir and is thus independent of any debris that may be produced in the test gearbox. High-speed elastomeric flex couplings and shafts connect the reaction gearbox to the test gearbox through the center support bearings.

The test gearbox houses the test gears and bearings. The test gearbox gear mesh and bearings are lubricated from the same oil reservoir. However, the oil supplied to the test gears can be controlled at a higher temperature up to 150 °C with an independent flow rate from 0 to 4 LPM. The test gears are constrained to a 1:1 ratio and 3.5-inch center distance, as seen in Figure 5. These parameters were chosen in accordance with the standard NASA test gear that has been used for decades of gear research. More information on the test gears is provided in Section 3.

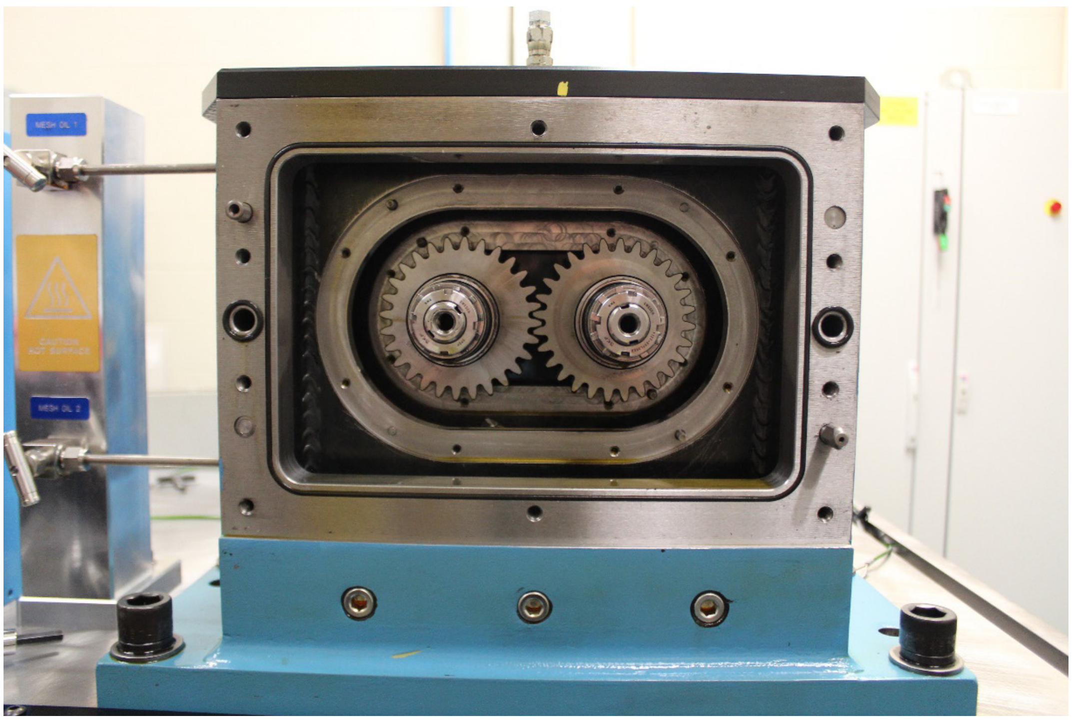


Figure 5. 1:1 gear mesh of FSG.

The test gears are supported by two test gear shaft assemblies (Figure 6). Each gear shaft can be removed from the gearbox and assembled on a bench. The assembly contains a bearing race, spacers, and test gear. The shafts are splined and the gears are held in place with a nut and washer lock system. The spacers can be adjusted to accommodate a gear of up to 0.787 inch (20 mm) face width. The gear shaft assembly allows quick removal and inspection of the test gears while also ensuring correct alignment when reassembling. Alignment is crucial for controlled and repeatable experiments.

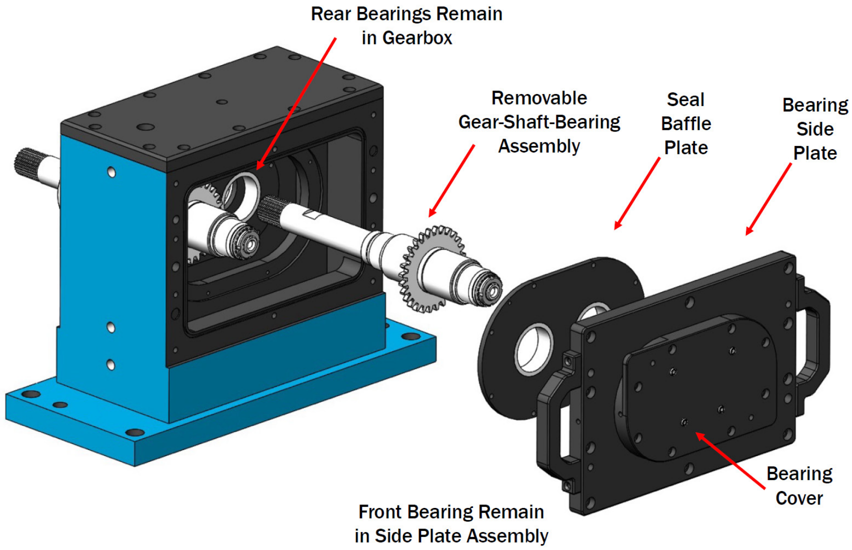


Figure 6. Test gearbox showing removable bearing cover and gear shaft assemblies.

2.3 Loss of Lubrication Capabilities

The FSG was uniquely designed for loss-of-lubrication (LoL) experiments. The Army is interested in developing new technologies to increase operating time and performance after an LoL condition to increase survivability in emergency situations.

The test gearbox has a three-chamber design (Figure 7). The chambers allow for isolation of the gear mesh by installing baffle plates and zero-clearance composite seals along the shafts, as highlighted in Figure 8. During an LoL test, the oil flow to the test gear mesh is turned off while the two outer chambers allow the support bearings to remain lubricated. The plates and seals ensure that no oil mist from the bearings enters the gear mesh, while also preventing debris from the gears from reaching the bearings. The three-chamber design provides the most severe LoL condition possible to the test gears and increases repeatability by eliminating the effect of uncontrolled bearing oil mist on the gear surfaces.

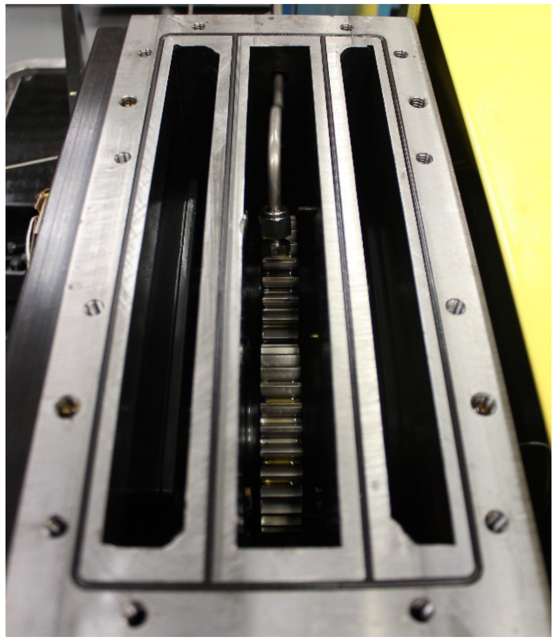
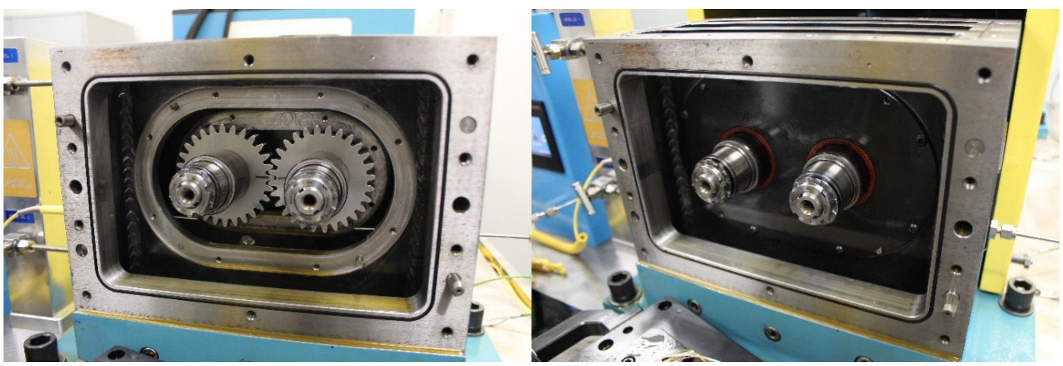


Figure 7. Top view of test gearbox showing three-chambered design, with bearings contained in the outer chambers and the test gear mesh in the center chamber.



(a) (b)

Figure 8. (a) Front view of test gearbox without baffle plates installed and (b) with baffle plates installed to isolate front bearing lubrication from reaching the test gear mesh.

2.4 Debris Injection Capabilities

The FSG includes a debris injection system enabling research on the impact of various debris to a gear mesh. Debris is added to a dry debris cylinder. The device uses an auger with a stepper motor to accurately add debris into the mesh oil via a debris shot. Figure 9 illustrates the debris injector concept. Baffles and composite seals prevent debris from reaching the bearings, and the test gearbox drain oil is directed to a dirty oil collection reservoir and filtering system. Debris of current interest includes additive manufacturing laser powder bed fusion powder

(20–80 µm), direct energy deposition powder (50–150 µm), sand (80–500 µm), and artificial test dust (0–250 µm).

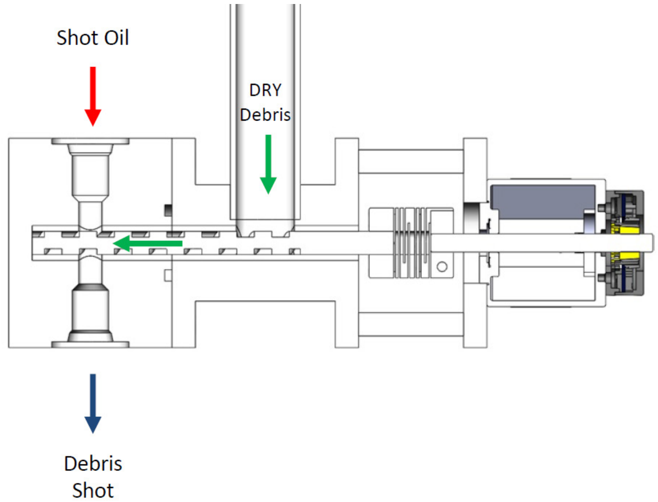


Figure 9. Debris injector schematic. An auger and motor are used to quantitatively add debris to the gear oil mesh.

2.5 Machine Controls

The rig is controlled by a separate human–machine interface (HMI) located away from the machine. The software allows automated experiment scripts to be created so that no real-time user input is needed to advance through desired parameters and conditions. The machine controls include automated health monitoring and shutdown limits to protect the rig in case conditions outside of the expected speed, torque, temperature, or vibration ranges are experienced. The rig also includes a manual emergency stop, and ability for a digital input/output shutdown signal to be incorporated from a third-party data acquisition and control setup. Maintenance safety lockouts including guard proximity sensors and HMI lockouts prevent the rig from rotating during maintenance.

3. FSG Test Gears

3.1 Test Gears

The test and reaction gearboxes were designed for a fixed 1:1 test gear ratio with a 3.5-inch center distance. This decision was made to follow the standard design used by NASA for decades of gear research. NASA used this design to investigate various gear technologies and experimental protocols such as loss of lubrication[4] and pitting fatigue life.[5] The design also shares many similarities with the gears used for the Ryder Rig, which is a gear rig used to assess the load capacity of lubricants. Lubricants assessed on the Ryder Rig include those used in rotorcraft transmissions.[6]

The test gear is a spur gear with 28 teeth, 0.75-inch hub, and 0.25-inch face width. It features a diametral pitch of 8 and a pressure angle of 20°. Key parameters of the test gear with dimensions are shown in Figure 10.

The FSG will use the standard test gear with a face width of 0.25 inch as well as a modified gear in which the face width is increased to the full hub width of 0.75 inch. The standard width gear is used for direct comparisons to lower torque experimental datasets, particularly historical data for pitting and loss of lubrication. The full hub width gear will allow the rig to be run at higher torques without immediate risk of tooth fracture. A complete drawing of the test gear is provided in Appendix B. These designs, and others, will be used to simulate specific gear contact conditions and produce desired failure modes based on the research project.

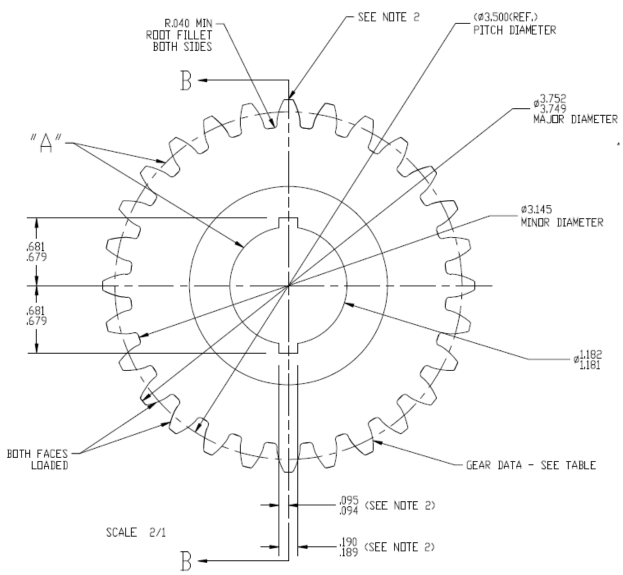


Figure 10. Test gear drawing. Design based on the NASA test gear.

3.2 Gear Calculations

Gear operating conditions, including sliding and entrainment velocities and bending stress for the FSG at maximum operating conditions, are calculated and provided below. All variables are defined in Appendix C. Full derivations of the equations can be found in *Shigley's Mechanical Engineering Design*[7] and ANSI/AGMA 2001-D04,[8] and *Simulating Army-Relevant Spur Gear Contacts with a Ball-on-Disc Tribometer*.[1]

Gear tooth meshing can be separated into five stages: 1) engagement, 2) rolling/sliding contact, 3) pure rolling contact, 4) rolling/sliding contact, and

5) disengagement. Figure 11 is a diagram of the meshing region of two gears, with line segment “SO” being the involute tooth profile. This diagram is used for references to gear mesh locations, line segment definitions, and angle definitions in the subsequent sections.

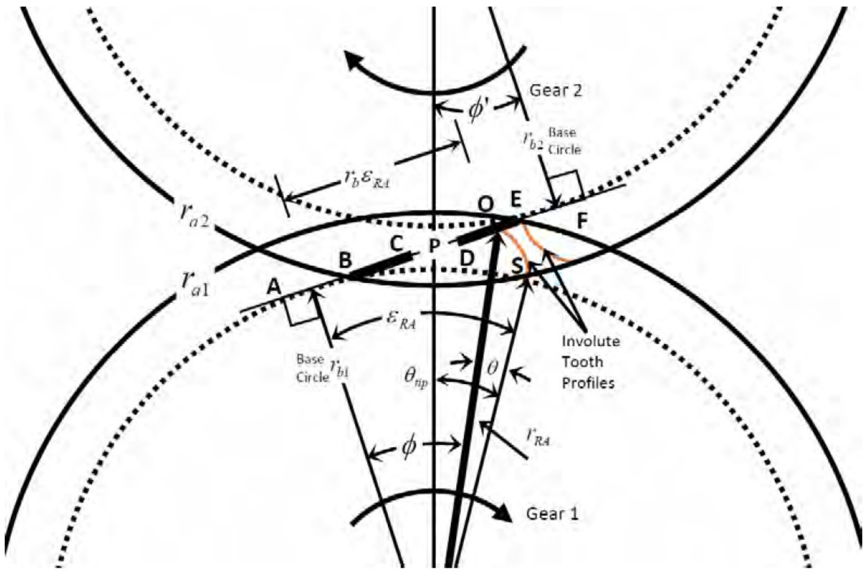


Figure 11. Gear tooth involute profile locations.[1]

3.2.1 Sliding and Entrainment Velocity

The sliding velocity, U_s, is defined in Equation 1 as the relative velocity among the meshing surfaces.

$$U_s = (\overrightarrow{U_1} - \overrightarrow{U_2}) \cdot \vec{T}, \tag{1}$$

where T is the tangent vector to the meshing surfaces. Using Equation 2

$$U_s = -(\omega_1 + \omega_2)\overline{PO} \tag{2}$$

as derived by Riggs et al.[1] and Figure 10, sliding velocities were calculated at several points along the pressure line.

Entrainment velocity is an average surface velocity in the sliding direction and is given by Equation 3:

$$U_e = \frac{1}{2}(\overrightarrow{U_1} + \overrightarrow{U_2}) \cdot \vec{T} \tag{3}$$

or as derived by Riggs et al.[1] simplifies to Equation 4:

$$U_e = -\frac{\omega_1}{2}\left\{\left(1 - \frac{r_{p1}}{r_{p2}}\right)\overline{PO} + 2\overline{AP}\right\} \tag{4}$$

3.2.2 Bending Stress

Bending stress can be calculated as the Lewis Bending Stress using Equation 5, taken from Shigley.[7] W^t is the tangential transmitted load, P is the diametral pitch, F is the face width of the narrowest member, and Y is the Lewis Form Factor.

$$\sigma = \frac{W^t P}{F Y} \tag{5}$$

Bending stress can also be calculated from American National Standards Institute-American Gear Manufacturers Association (ANSI-AGMA) 2001-D04,[8] Equation 6, in which K_o, K_v, K_s, K_m, and K_b and J are AGMA correction factors. These factors account for transmitted load, overload, dynamic effects, size, geometry, distribution of load, rim support, and stress concentrations.[7]

$$s_t = W_t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J} \tag{6}$$

3.2.3 Contact Stress

Hertzian contact stress is given by Equation 7, taken from Shigley,[7]

$$\sigma_C^2 = \frac{W^t}{\pi F \cos\phi} \frac{(1/r_1) + (1/r_2)}{\left[\left(1 - \nu_1^2\right)/E_1\right] + \left[\left(1 - \nu_2^2\right)/E_2\right]} \tag{7}$$

where

$$r_1 = \frac{d_P \sin\phi}{2} \qquad r_2 = \frac{d_G \sin\phi}{2} \tag{8,9}$$

where r_1 and r_2 are the instantaneous values of radii of curvature on the pinion and gear, ϕ is the pressure angle, d_p and d_G, are the pitch diameters of the pinion and gear, respectively, and E_1, E_2, v_1, v_2 are the Poisson's ratio and Modulus of Elasticity of the gear materials.

Contact stress can also be calculated using ANSI-AGMA 2001-D04,[8] Equation 10 below,

$$s_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{d F} \frac{C_f}{I}} \tag{10}$$

where Cp defined by AGMA as the elastic coefficient[7] and is calculated by

$$C_p = \sqrt{\frac{1}{\pi\left[\left(\frac{1-\mu_P^2}{E_P}\right)+\left(\frac{1-\mu_G^2}{E_G}\right)\right]}} \tag{11}$$

K_o, K_v, K_s, K_m, C_f, and I are AGMA correction factors and μ_P, μ_G, E_P, and E_G are the Poisson's ratio and Modulus of Elasticity of the gear materials.

Using the 0.75-inch-face-width gear and the FSG at maximum speed of 12,500 rpm and maximum torque of 332 ft-lb (450 Nm), the following conditions are produced (Table 1).

Table 1. FSG operating conditions at maximum speed and torque.

Gear mesh parameter	Metric	Imperial
Sliding velocity	20.10 m/s	791.3 inch/s
Entrainment velocity	19.9 m/s	783.5 inch/s
Bending stress (AGMA)	0.62 GPa	89.9 ksi
Contact stress (AGMA)	1.88 GPa	272.7 ksi

Maximum contact stresses of 1.5 GPa and bending stresses of 0.4 GPa are rarely exceeded in typical Army helicopters.[1,9] The FSG exceeds these conditions and allows margins to improve materials and increase power to weight ratios of future rotorcraft. Additional calculations for various FSG operating speeds and torques are provided in Appendix D.

4. Conclusion

The FSG is a U.S. Army research asset that will enable gear research and development through highly controlled and repeatable experiments. Its custom design for loss of lubrication and debris injection will provide unique research capabilities. The FSG can meet and exceed current Army rotorcraft gear loads allowing for gear technologies to be evaluated at conditions directly found in rotorcraft. The FSG provides component-level experimental capabilities to develop and evaluate new and improved gear technologies.

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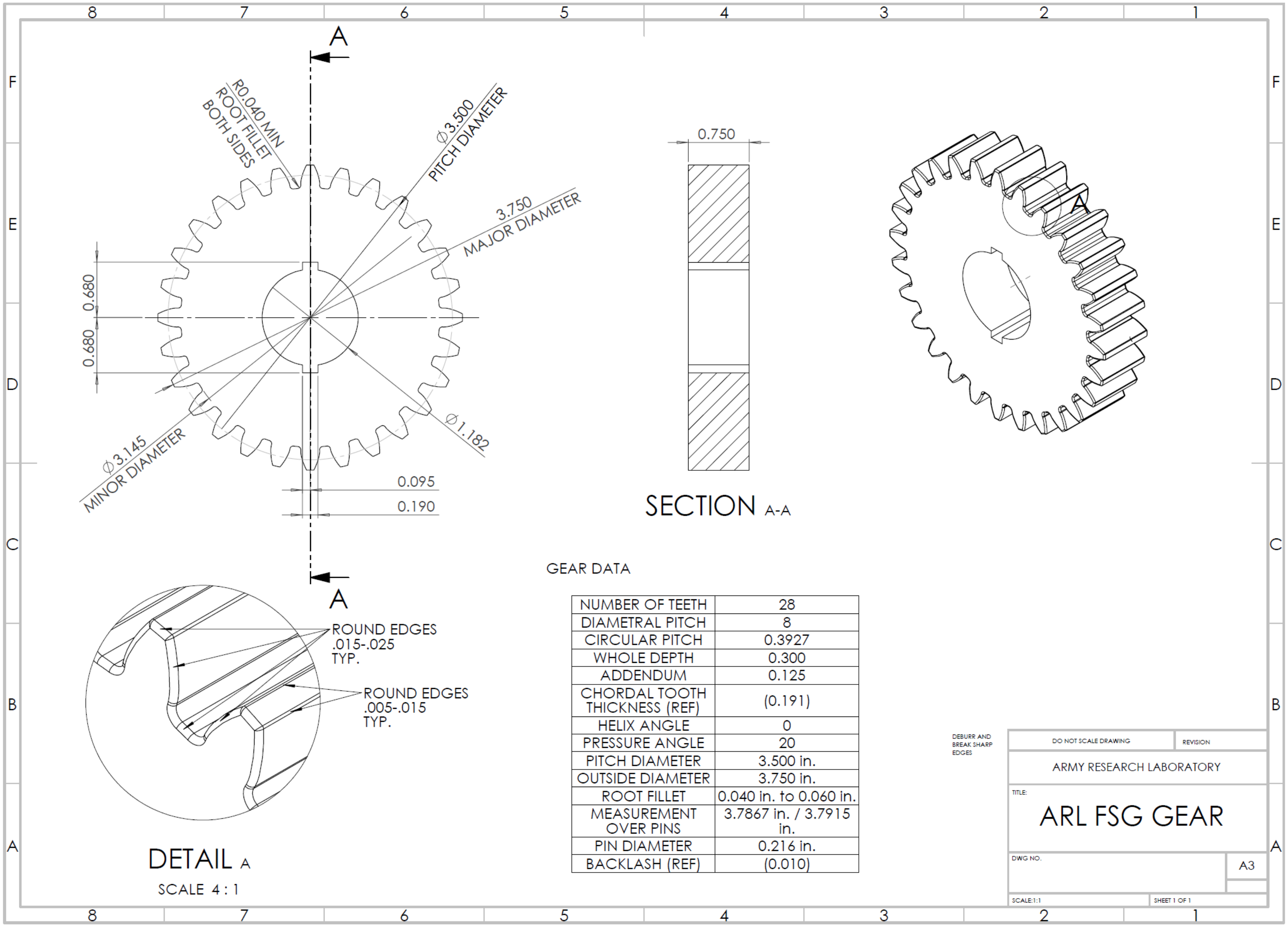
Appendix A. Four-Square Gear Rig (FSG) Technical Specifications

Table A-1. FSG technical specifications.

Machine parameter	Value
Test gear center distance	3.5 inches
Test gear ratio	1:1
Minium gear speed	1,000 rpm (Bi-directional)
Maximum gear speed	12,500 rpm (Bi-directional)
Minimum gear torque	25 Nm (Bi-directional)
Maximum gear torque	450 Nm (Bi-directional)
Oil capacity	30 L
Oil temperature	40 to 150 °C
Test gear oil flowrate	0–4 LPM

Appendix B. Four-Square Gear Rig (FSG) Gear Drawing; 0.75-inch Face Width



NUMBER OF TEETH	28
DIAMETRAL PITCH	8
CIRCULAR PITCH	0.3927
WHOLE DEPTH	0.300
ADDENDUM	0.125
CHORDAL TOOTH THICKNESS (REF)	(0.191)
HELIX ANGLE	0
PRESSURE ANGLE	20
PITCH DIAMETER	3.500 in.
OUTSIDE DIAMETER	3.750 in.
ROOT FILLET	0.040 in. to 0.060 in.
MEASUREMENT OVER PINS	3.7867 in. / 3.7915 in.
PIN DIAMETER	0.216 in.
BACKLASH (REF)	(0.010)

Figure B-1. FSG gear drawing; 0.75-inch face width.

Appendix C. Nomenclature Table

Table C-1. Nomenclature table.

Variable	Units	Description
U_s	inch/s	Sliding velocity
$\overrightarrow{U_1}$	inch/s	Velocity of surface no. 1
$\overrightarrow{U_2}$	inch/s	Velocity of surface no. 2
$\vec{T}$	none	Tangent vector to the meshing surfaces
ω_1	rad/s	Angular velocity of gear no. 1
ω_2	rad/s	Angular velocity of gear no. 2
$\overline{PO}$	inch	Length of line segment
U_e	inch/s	Entrainment velocity
r_{p1}	inch	Pitch radius of gear no. 1
r_{p2}	inch	Pitch radius of gear no. 2
$\overline{AP}$	inch	Length of line segment
σ	psi	Bending stress
W^t	lb	Transmitted load
P	Teeth/inch	Diametral pitch
F	inch	Face width of narrowest gear
Y	. . .	Lewis form factor
K_o	. . .	Overload factor
K_v	. . .	Dynamic factor
K_s	. . .	Size factor
K_m	. . .	Load distribution factor
K_b	. . .	Rim thickness factor
s_t	psi	Bending stress, AGMA definition
W_t	lb	Transmitted load
P_d	inch^{-1}	Transverse diametral pitch
J	. . .	Geometry factor for bending strength
σ_c	psi	Hertzian contact stress
φ	deg	Pressure angle
r_1	inch	Radius of curvature of tooth profile at pitch point
r_2	inch	Radius of curvature of tooth profile at pitch point
d_p	inch	Diameter of pinion
d_g	inch	Diameter of gear
s_c	psi	Contact stress
C_p	$[\text{lb/inch}^2]^{0.5}$	Elastic coefficient
C_f	. . .	Surface condition factor for pitting resistance
I	. . .	Geometry factory for pitting resistance
μ_G	. . .	Poisson's ratio for gear
μ_P	. . .	Poisson's ratio for pinion
E_G	lb/inch^2	Modulus of elasticity for gear
E_P	lb/inch^2	Modulus of elasticity for pinion

Appendix D. Four-Square Gear Rig (FSG) Operating Conditions

This appendix contains tables of gear meshing information for three sample operating conditions and gear face widths. Five values are given for the roll angle, sliding velocity, and entrainment velocity. These correspond to points B, C, P, D, and E along the driving tooth.

Table D-1. FSG gear mesh information for condition 1 (maximum power).

FSG condition 1 (maximum power)					
Designation	**Spur gear**				
Speed (rpm)	12,500				
Torque (ft-lb)	332				
Power (Hp)	790				
No. teeth	28				
Diametral pitch	8				
Pressure angle (°)	20				
Pitch diameter (inch)	3.5				
Outside diameter (inch)	3.75				
Face width (inch)	0.75				
Points	B	C	P	D	E
Roll angles (°)	10.32	18.53	20.85	23.18	31.38
Sliding velocity (inch/s)	–791.22	–174.85	0	174.85	791.22
Entrainment velocity (inch/s)	783.48	783.48	783.48	783.48	783.48
Contact stress (ksi)	272.76				
Bending stress (ksi)	90.62				

Table D-2. FSG gear mesh information for condition 2 (medium power).

FSG condition 2 (medium power)					
Designation	**Spur gear**				
Speed (rpm)	10,000				
Torque (ft-lb)	250				
Power (Hp)	476				
No. teeth	28				
Diametral pitch	8				
Pressure angle (°)	20				
Pitch diameter (inch)	3.5				
Outside diameter (inch)	3.75				
Face width (inch)	0.75				
Points	B	C	P	D	E
Roll angles (°)	10.32	18.53	20.85	23.18	31.38
Sliding velocity (inch/s)	–632.98	–139.89	0	139.89	632.98
Entrainment velocity (inch/s)	626.78	626.78	626.78	626.78	626.78
Contact stress (ksi)	234.99				
Bending stress (ksi)	67.27				

Table D-3. FSG gear mesh information for condition 3 (NASA speed and torque condition[1]).

FSG condition 3 (NASA speed and torque condition)					
Designation			**Spur gear**		
Speed (rpm)			10,000		
Torque (ft-lb)			43.74		
Power (Hp)			83.3		
No. teeth			28		
Diametral pitch			8		
Pressure angle (°)			20		
Pitch diameter (inch)			3.5		
Outside diameter (inch)			3.75		
Face width (inch)			0.25		
Points	B	C	P	D	E
Roll angles (°)	10.32	18.53	20.85	23.18	31.38
Sliding velocity (inch/s)	–632.98	–139.89	0	139.89	632.98
Entrainment velocity (inch/s)	626.78	626.78	626.78	626.78	626.78
Contact stress (ksi)			164.01		
Bending stress (ksi)			32.76		

[1] Handschuh RF, Gargano LJ. Test facility simulation results for aerospace loss-of-lubrication of spur gears. National Aeronautics and Space Administration (US); 2014 Oct. Report No.: NASA/TM-2016-218396.

List of Symbols, Abbreviations, and Acronyms

AGMA	American Gear Manufacturers Association
ANSI	American National Standards Institute
ARL	Army Research Laboratory
DEVCOM	U.S. Army Combat Capabilities Development Command
FSG	Four Square Gear Rig
HMI	Human–Machine Interface
LoL	Loss of Lubrication
NASA	National Aeronautics and Space Administration
rpm	Revolutions per Minute
VIPER	Vehicle Innovative Powertrain Experimental Research

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